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Modelling framework for integration of large-scale heat pumps in district heating using low-temperature heat sources: A case study of Tallinn, Estonia

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Abstract
The paper presents a modelling framework that may be used to plan the integration of large-scale HPs in district heating (DH) areas. By use of the methodology both optimal HP capacities to be installed and optimal choice of heat source to be used during the year are identified by minimizing total cost of ownership including investment and operational costs. The modelling framework uses mixed-integer linear programming and hourly calculations over one year. Seasonal variations of the heat source temperatures, capacity limitations and HP coefficient of performance as well as technical constraints were taken into account.
The DH network of Tallinn, Estonia, was used as a case study. Six different heat source types were identified for 13 potential locations of large-scale HPs.
The results showed that the integration of large-scale HPs in the DH network of Tallinn is economically feasible. It was found that 122 MW HP capacity could be installed without compromising the operation of sustainable base load units. The heat sources needed for obtaining this solution were sewage water, river water, ambient air, seawater and groundwater. It was further shown that the Lorenz efficiency depends on the variations of heat source temperatures.

Keywords
District heating;
Energy planning;
Large-scale heat pumps;
Low-temperature heat sources;
Optimization;

1. *Introduction*

Large-scale heat pumps (HPs) have been identified as an important technology to utilize intermittent power production from renewable energy sources (RES) by integrating the power and heating sectors [1,2]. Energy planning tools are often used in order to investigate the integration possibilities of large-scale HPs in district heating (DH) networks directly or to evaluate how a high share of renewable energy sources (RES) can be integrated in energy systems by the use of large-scale HPs. Integrating a detailed thermodynamic model of a HP in energy planning tools is complicated and may make the optimization problem very complex to solve. Accordingly, HPs are often represented in a simplified way, e.g. by assuming a constant coefficient of performance (COP) [3–8] or by adjusting the COP with changes in heat demand [9].
The COP of a HP, however, depends on many factors. The choice of refrigerant and equipment, such as compressor and/or heat exchanger type, as well as operational heat source and heat sink temperatures affect the COP. While component specific decisions are often not made at the planning stage, heat source and heat sink temperatures for the investigated area may be known. Since the heat sink temperatures, in form of DH supply and return temperature, vary over the year, the COP of a HP would also change. In addition, heat sources such as ambient air and surface water (seawater, lakes and rivers) also vary in temperature over the year. Consequently, the use of a constant COP of HPs using renewable heat sources to supply DH networks is not adequate. Therefore, it may be important for energy planning to calculate the COP based on a daily or hourly time step.
Such effects of varying heat source and heat sink temperatures were taken into account by studies from Lund et al. [10] and Østergaard and Andersen [11], who related COP to an ideal Lorenz cycle with a constant Lorenz efficiency of 0.4 and 0.5, respectively. A Lorenz cycle considers the inlet and outlet temperatures of the heat source and the heat sink of a HP. Lund et al. [10] used the energy system analysis tool EnergyPlan [12], which

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Nomenclature

C	Costs, €	e	Expansion
CF	Discounted cash flows, €	el	Electricity
c_p	Specific heat, MJ/kg/K	H	Heat sink
COP	Coefficient of performance, -	HP	Heat pump
f_Q	Compressor heat loss, -	i	Inlet
L	Lifetime of plant, yrs	Inv	Investment
l	Index for current year	is	isentropic
LCOH	Levelized cost of heat, €	o	Outlet
NPV	Net present value, €	L	Lorenz cycle
P	Electricity, MWh	lift	Temperature lift of heat pump
PBT	Payback time, yrs	m	Operating and maintenance
Q	Heat, MWh	min	minimum
$\dot{Q}$	Heat capacity, MWh/h	NG	Natural gas
R	Revenue, €	p	Production plant
r	Discount rate, -	pp	Pinch point
T	Temperature, °C	r	Refrigerant
TCI	Total capital investment, €	rural	Rural area
u	Binary variable, -	t	hour
$\dot{V}$	Volume flow rate, m^3/s	TLL	Tallinn
$\dot{W}$, w	Work, MW	v	variable
Z	Objective function, €		
		Abbreviations	
Greek symbols		CHP	Combined heat and power
$\Delta\bar{T}$	Temperature difference, K	COP	Coefficient of performance
η	Efficiency, -	DH	District heating
ρ	Density, kg/m^3	FLH	Full load hours
		HP(s)	Heat pump(s)
Subscript		LCOH	Levelized cost of heat
a	annual	NPV	Net present value
C	Heat source	O&M	Operating and maintenance
c	Compression	PBT	Payback time
d	Design conditions	RES	Renewable energy sources
DH	District heating	SCOP	Seasonal coefficient of performance

is a tool that has widely been used for studies on energy planning, e.g. in [13]. They focused on the analysis of low-temperature DH concepts at different DH network temperatures. Østergaard and Andersen [11] used the software energyPRO [14] to investigate the optimal use of central HPs and decentral booster HPs for an electrified low-temperature DH system based on RES. They focused on identifying optimal DH temperatures to increase COP of the HPs and to reduce heat losses.

Jensen et al. [15] derived a generic equation to calculate the COP and showed that the Lorenz efficiency of a HP varies for changes in the difference between the absolute mean temperatures of the heat source and the heat sink. The equation for calculating the COP was derived analytically, using a number of terms and factors, which may be based on typically known parameters and reasonable assumptions. Approximations were given for those factors that are more difficult to estimate without a thermodynamic HP model.

1.1. Heat sources used for HPs

Information about available heat sources, their temperature variations and capacity limitations over the year are often not known or not further specified, as in the studies which assumed a constant COP [3–8]. They either used typical values for COP or yearly average values. Østergaard and Andersen [11] assumed a constant heat source temperature. Lund et al. [10] used measured seawater temperatures and added a constant temperature to this profile to represent other heat sources, such as sewage water. A few studies have investigated different heat sources used for HPs in more detail.

Gaudard et al. [16] investigated the potential of using lakes and rivers in Switzerland as heat source and heat sink for heating and cooling purposes. They used a hydrodynamic model and considered seasonal variations of the lakes and rivers in order to calculate the potential capacities of rivers and lakes, which they mapped and compared with the local heating and cooling demands.

Lund and Persson [17] analysed available heat sources, which could be used by HPs to supply DH in Denmark. They mapped the heating capacity of available heat sources together with the heat demands.

For Sweden, Berntsson [18] analysed the most commonly used heat sources for HPs in individual buildings and for large systems connected to DH. They looked at annual temperature variations, temperature levels and availability of the different heat sources. In addition, economic and environmental parameters were analysed. The main heat sources for individual buildings were ambient air, exhaust air, lake or river water, soil and rock. Lake water and cleaned sewage water were predominantly used for large HP systems.

David et al. [19] reviewed and analysed existing large-scale HPs in DH systems in Europe. They identified 149 installed large-scale HPs by 2017 with a total capacity of 1580 MW. The different heat sources, which were used the most were sewage water, surface water (seawater, lakes and rivers), industrial waste heat and geothermal water. They concluded that sewage water is the most suitable heat source for HPs considering temperature, long-term stability and shortest distance to urban areas. Surface water may also result in good performance.

These studies analysed the potentials of using certain heat sources to supply heat by HPs and DH. The most suitable heat sources were identified for a certain region and also the potential of the heat source in terms of temperature and capacity were compared to heat demands.

1.2. HP investment costs

Investment costs of large-scale HP projects are not well established. Wolf et al. [20] developed cost correlations for HPs with a thermal capacity of up to 0.2 MW for ground-source, water-source and air-source HPs. Grosse et al. [21] developed a correlation for large-scale HPs based on reference project information and estimated offers from manufacturers. However, all of these correlations are only valid for the HP unit itself. Investment costs related to the heat source, planning of the project, grid connection, control or the building were not further specified. The Danish Energy Agency estimated the total specific investment costs for large-scale HP projects to be between 0.8 million €/MW and 1.1 million €/MW [22].

Pieper et al. [23] analysed existing and planned large-scale HP projects in Denmark, as well as a number of offers for specific HP units. They allocated the costs to the HP unit itself, the heat source, construction, consulting and electricity related costs. Cost correlations were developed for five different heat sources: ambient air, groundwater, sewage water, industrial excess heat and flue gas. They further stated that the HP itself contributed with only 38 % to 54 % of the overall investment of the investigated projects. Furthermore, the specific investment costs depended on the installed capacity, because some costs of the projects depended on the size of HP, while others did not.

1.3. The Estonian case

In year 2000, Lund et al. [24] proposed to change the Estonian energy system from oil-shale based electricity production units to cogeneration plants. This would reduce the primary energy consumption. It was suggested to install combined heat and power (CHP) plants based on biomass, because Estonia had forest coverage of 45 %. This would result in a sustainable supply of power and heat without requiring imports of biomass. In addition, the existing DH infrastructure would have to be renovated. The strategy proposal has been pursued, as for instance the capital Tallinn has installed two new biomass-based and one waste incineration CHP plant over the last decade, which supplied 50 % of DH in 2017 [25].

Blumberga et al. [26] analysed how the power system in the Baltic States would have to change until 2050 in order to fulfil the political targets of becoming more sustainable. It was found that Estonia would have to decrease the capacity of oil-shale-based electricity generation plants and increase the share of biomass CHP plants, as well as on- and offshore wind turbines in order to fulfil its goals.

Lauka et al. [27] investigated the integration possibilities of HPs into DH for the Baltic States with Latvia as example. They conclude that HPs would increase the electricity demand and thereby be able to help integrating more RES.

1.4. Scope

The literature survey shows that models accounting better for investment costs of large-scale HPs, variations in the COP of the HPs, as well as availability, capacity and temperature of the heat sources may be useful.

Since it is important to know how these aspects impact on the decision making about the integration of large-scale HPs in existing or new DH areas, we propose a new modelling framework in this study.
The purpose of the model was to identify how large-scale HPs can be efficiently integrated in DH areas by addressing all of the above mentioned aspects reflecting the local conditions and yearly variations. The developed model was applied to the city of Tallinn as a case study with the aim to identify:

- the most suitable heat sources to be used by large-scale HPs
- the optimum HP capacity to be installed for each heat source
- optimal hourly operation of the chosen HPs to minimize costs

The model takes investment costs, operating and maintenance (O&M) costs, seasonal temperature variations of heat sources and heat sink, capacity limitations of the heat sources as well as the distance from the heat sources to the DH network into account.

2. *Methods*

The formulation of the model including the objective function, important HP approximations and performance indicators are presented in the following subsections.

2.1. Model formulation

The software GAMS, version 24.8.3, [28] was used to perform an optimization based on mixed-integer linear programming using the CPLEX solver, version 12.7.0.0 [29]. The aim of the optimization was to minimize total costs, including annualized investments and annual O&M costs of all production plants *p*, as shown in Eq. (1). The revenue of selling the produced heat from the HPs, $R_{HP,p}$, was deducted, so that the objective function resulted in a negative value, representing the profit. The time step for the calculations was one hour. The revenue of heat sales was limited by the levelized cost of heat of the current production unit, which will be replaced by a HP. This represents the maximum price the HP operator could achieve compared to the current production unit.

$$\min Z = \sum_p Z_p = \sum_p C_{\mathrm{el},a,p} + C_{m,a,p} + C_{\mathrm{HP},a,p} + C_{\mathrm{DH},a,p} - \mathrm{R}_{HP,p} \tag{1}$$

Investment costs for the HPs C_{HP} and DH piping C_{DH} were annualized to reduce calculation time, as described in [30] and shown in Eq. (2), considering the lifetime of the technology *L* and discount rate *r*. For DH pipes, the investment costs until up to the lifetime of the HP were considered.

$$C_{\mathrm{Inv},a} = C_{\mathrm{Inv}} \frac{r}{(1+r)(1-(1+r)^{-L})} \tag{2}$$

Investment costs of large-scale HPs, distinguished by the type of heat source used, were obtained from Pieper et al. [23], which consisted of a constant parameter $c_{p,\min}$ and a parameter $c_{p,v}$, which was multiplied by the installed HP capacity $\dot{Q}_p$. A general form is shown in Eq. (3), where u_p is a binary variable defining whether a HP will be installed or not. The optimal HP capacity was found by the optimization.

$$C_p = c_{p,\min} u_p + c_{p,v} \dot{Q}_p \tag{3}$$

The DH piping costs were formulated in a similar way with a constant parameter and one term depending on the pipe diameter, consequently also on the capacity. Annual O&M costs $C_{m,a,p}$ were taken from [31] and consisted of a parameter depending on the installed HP capacity and one on the supplied heat.
The annual costs for electricity consumption of the HPs $C_{\mathrm{el},a,p}$ were based on the day-ahead hourly electricity price, taxes and tariffs. Value-added tax was omitted.
The hourly heat supplied by all HPs ($Q_{\mathrm{HP},t}$) could not be larger than the heat from the technology the HPs replace $Q_{NG,t}$ and each HP capacity $\dot{Q}_{\mathrm{HP},p}$ was always larger than any hourly heat supply by that HP.

$$Q_{\mathrm{HP},t} = \sum_p Q_{\mathrm{HP},p,t} \leq Q_{\mathrm{NG},t} \tag{4}$$

$$Q_{\mathrm{HP},p,t} \leq \dot{Q}_{\mathrm{HP},p} \tag{5}$$

The relation between the heat production and the consumed electricity of a HP is defined as the COP:

$$\mathrm{COP} = \frac{Q_{\mathrm{HP}}}{P_{\mathrm{HP}}} \tag{6}$$

Therefore, the HP with the highest COP would lead to the lowest electricity consumption. In addition, constraints of the heat source were considered by limiting the HP capacity or the heat source volume flow rate, due to limited availability of the heat source.

2.2. Calculation of HP COP

The COP of a HP was modelled by an approach presented by Jensen et al. [15], who derived a generic equation for the COP analytically for a single stage HP cycle, which is shown in Eq. (7).

$$\mathrm{COP} = \left(\mathrm{COP}_L \frac{1 + \frac{\Delta\bar{T}_{r,H} + \Delta\bar{T}_{\mathrm{pp}}}{\bar{T}_H}}{1 + \frac{\Delta\bar{T}_{r,H} + \Delta\bar{T}_{r,C} + 2\Delta\bar{T}_{\mathrm{pp}}}{\Delta\bar{T}_{\mathrm{lift}}}} \eta_{\mathrm{is},c} \left(1 - \frac{w_{\mathrm{is},e}}{w_{\mathrm{is},c}}\right) + 1 - \eta_{\mathrm{is},c} - f_Q \right) \tag{7}$$

The equation was derived for estimation of the performance of a HP cycle in design conditions. However, it was found that it gives reasonable results also for a two-stage HP model running at full capacity with annual variations in temperatures using a constant correction factor of 1.05, see Appendix 1. This correction factor fits in values with typical performance improvements from selecting one or two-stage HPs [32]. Using the proposed equation, the COP depends only on characteristics of the heat source and heat sink temperatures (COP_L, $\bar{T}_H$ and $\Delta\bar{T}_{\mathrm{lift}}$) as well as characteristics of the compressor ($\eta_{\mathrm{is},c}$ and f_Q), the heat exchangers ($\Delta\bar{T}_{\mathrm{pp}}$) and certain characteristics of the refrigerant ($w_{\mathrm{is},e}/w_{\mathrm{is},c}$, $\Delta\bar{T}_{r,H}$ and $\Delta\bar{T}_{r,C}$). Approximations were proposed by Jensen et. al [15] for $\Delta\bar{T}_{r,H}$, $w_{\mathrm{is},e}/w_{\mathrm{is},c}$ and $\Delta\bar{T}_{r,C}$ using ammonia as the refrigerant. Ammonia is a refrigerant typically used in large-scale HPs supplying DH [31]. The input values and parameters based on inlet (*i*) and outlet (*o*) temperatures of heat source (*C*) and heat sink (*H*) may be found in Table 1 and Eq. (8).

Parameter	Symbol	Value	Unit
Correction factor		1.05	-
Compressor heat loss	f_Q	0.05	-
Compressor isentropic efficiency	$\eta_{\mathrm{is},c}$	0.80	-
Entropic average pinch point temperature difference	$\Delta\bar{T}_{\mathrm{pp}}$	5	K
Refrigerant induced temperature difference at evaporator	$\Delta\bar{T}_{r,C}$	approx. from [15]	K
COP of a Lorenz cycle	COP_L	$\bar{T}_H/(\bar{T}_H - \bar{T}_C)$	-
Temperature lift the HP has to overcome	$\Delta\bar{T}_{\mathrm{lift}}$	$\bar{T}_H - \bar{T}_C$	K
Refrigerant induced temperature difference at condenser	$\Delta\bar{T}_{r,H}$	approx. from [15]	K
Ratio of isentropic expansion and compression	$w_{\mathrm{is},e}/w_{\mathrm{is},c}$	approx. from [15]	-

Table 1: Input parameters for COP estimation (based on [15])

$$\bar{T}_H = \frac{T_{H,o} - T_{H,i}}{ln(T_{H,o}) - ln(T_{H,i})}, \qquad \bar{T}_C = \frac{T_{C,i} - T_{C,o}}{ln(T_{C,i}) - ln(T_{C,o})} \tag{8}$$

2.3. Design conditions of HPs

Design conditions for every HP using each heat source were defined for dimensioning purposes. Design conditions were represented by a typical winter day with lowest and highest occurring temperatures during the year for heat source and heat sink, respectively. Specifying inlet and outlet temperatures for every heat source and the heat sink allowed calculating COPs in design conditions and determining the heat source capacities $\dot{Q}_{d,\mathrm{source},p}$, as shown in Eq. (9), assuming no compressor heat losses and steady state energy balance for the HP [33].

$$\mathrm{COP}_{d,p} = \frac{\dot{Q}_{d,H}}{\dot{W}_{\mathrm{HP},p}} = \frac{\dot{Q}_{d,H}}{\dot{Q}_{d,H} - \dot{Q}_{d,C,p}} \Rightarrow \dot{Q}_{d,C,p} = \frac{\dot{Q}_{d,H}}{\mathrm{COP}_{d,p}}\left(\mathrm{COP}_{d,p} - 1\right) \tag{9}$$

The design volume flow rate of the heat source side was calculated considering the density and specific heat, as shown in Eq. (10).

$$\dot{V}_{d,C,p} = \frac{\dot{Q}_{d,\mathrm{C},p}}{\rho_{d,C,p} c_{p,d,p}\left(T_{C,i,d,p} - T_{C,o,d,p}\right)} \tag{10}$$

Consequently, the heat source capacity for every hour *t* was calculated based on the design volume flow rate using Eq. (11). The maximum and minimum volume flow rate of refrigerant also results in limitations to capacity, which was not included in this study.

$$\dot{Q}_{C,p,t} = \dot{V}_{d,C,p} \rho_{C,p,t} c_{p,p,t}\left(T_{C,i,p,t} - T_{C,o,p,t}\right) \tag{11}$$

Finally, the heat sink capacity limited by the volume flow rate was calculated for every hour based on Eq. (9), which further limited the hourly heat supply by the HPs, as shown in Eq. (12).

$$Q_{\mathrm{HP},p,t} \leq \dot{Q}_{H,p,t} \tag{12}$$

2.4. Performance indicators

Different performance indicators were used to evaluate the results from the optimization. Technical, economic and environmental indicators were considered.

2.4.1. Technical performance indicators

The annual hourly variation of the Lorenz efficiency η_L was determined for each heat source based on Eq. (13) considering the COP calculated by Eq. (7) and the Lorenz COP_L found in Table 1.

$$\eta_L = \frac{\mathrm{COP}}{\mathrm{COP}_L} \tag{13}$$

The required HP capacity $\dot{Q}_{\mathrm{HP},p}$ was determined by the maximum hourly heat produced by the HP with respect to the constraints described previously. The seasonal COP (SCOP) of each HP was calculated by dividing its annual heat production and annual electricity consumption. Thereby, the seasonal variation in heat demand and COP were taken into account when evaluating the individual units. The number of full load hours (FLH) was determined by dividing the annual heat production from each HP by its design capacity. Similar calculations of SCOP and FLH were also performed for the entire system, consisting of all HPs.

2.4.2. Economic performance indicators

The levelized costs of heat (LCOH) included all costs of producing the heat divided by the annual production of heat $Q_{a,p}$, as shown in Eq. (14). The sum over all HPs resulted in the total LCOH.

$$\mathrm{LCOH}_{\mathrm{HP},p} = \frac{C_{\mathrm{el},a,p} + C_{m,a,p} + C_{\mathrm{HP},a,p} + C_{\mathrm{DH},a,p}}{Q_{a,p}} \tag{14}$$

The annual production costs of the HP were calculated as the sum of the annual O&M costs and the costs for annual electricity consumption, including electricity taxes and tariffs, divided by the annual heat production:

$$\mathrm{Cost}_{\mathrm{el},m,p} = \frac{C_{\mathrm{el},a,p} + C_{m,a,p}}{Q_{a,p}} \tag{15}$$

The net present value (NPV) was calculated using the total capital investment (TCI) and discounted cash flows (CF), as shown in Eq. (16) [34]. The cash flows were calculated by considering the LCOH from the current production unit the HPs would replace, $\mathrm{LCOH}_{\mathrm{NG}}$. This would be the maximum price, R_{HP}, the HP operator could receive while being more competitive than the current production unit. The annual O&M costs of the HP operation and costs of annual electricity consumption were then deducted from that value resulting in the annual cash flow.

$$\mathrm{NPV} = -\mathrm{TCI} + \sum_{l=1}^{L} \frac{\mathrm{CF}_l}{(1+r)^l} = -C_{\mathrm{inv}} - C_{\mathrm{DH}} + \sum_{l=1}^{L} \frac{\mathrm{R}_{\mathrm{HP}} - C_{\mathrm{el},a} - C_{m,a}}{(1+r)^l} \tag{16}$$

The simple payback time (PBT) was calculated as shown in Eq. (17):

$$\mathrm{PBT} = \frac{\mathrm{TCI}}{\mathrm{CF}} = \frac{C_{\mathrm{inv}} + C_{\mathrm{DH}}}{\mathrm{R}_{\mathrm{HP}} - C_{\mathrm{el},a} - C_{m,a}} \tag{17}$$

2.4.3. Environmental performance indicator

The total generation of CO_2 by consumed electricity of the HPs was calculated by considering the total annual electricity consumption multiplied by the national annual CO_2 emission factor for electricity production. The equivalent amount of heat generated by the current production unit was used to calculate the amount of CO_2 emissions for that unit, considering the emission factor of the used fuel and the efficiency of that plant. The ratio between these two, the carbon-ratio, compared greenhouse gas emissions from producing heat with a HP and the current production unit.

3. *Case study: Tallinn DH*

The city of Tallinn, Estonia, was considered as case study in order to investigate the feasibility of implementing large-scale HPs for supply to the DH network. In Tallinn, 50 % of the heat demand in 2017 was considered fossil-free, because it was based on burning biomass and waste in CHP plants [25]. The remaining heat was supplied by five natural gas fired boilers. Currently, one more biomass based CHP plant is under construction, which will result in a production capacity of non-fossil fuels of 220 MW in 2019 [35]. The implementation of large-scale HPs may consequently be limited due to the already existing newly installed renewable production, but it could, on the other hand, result in a cheaper, more efficient and more sustainable heat supply than using natural gas fired boilers. Therefore, the potential for replacing heat production from natural gas boilers by large-scale HPs was investigated.

Supply temperature limitations of the HPs were considered by disregarding hours with supply temperatures above 85 °C, which is currently the limit for large-scale HPs that use ammonia as refrigerant [36]. This limitation could be addressed by considering various implementation schemes for large scale HPs in CHP systems [37]. Hourly measurements of supply and return temperature of the DH network in 2016 were used for the calculations [38]. The load duration curve and supply temperatures for the DH network of Tallinn are shown in Figure 1.

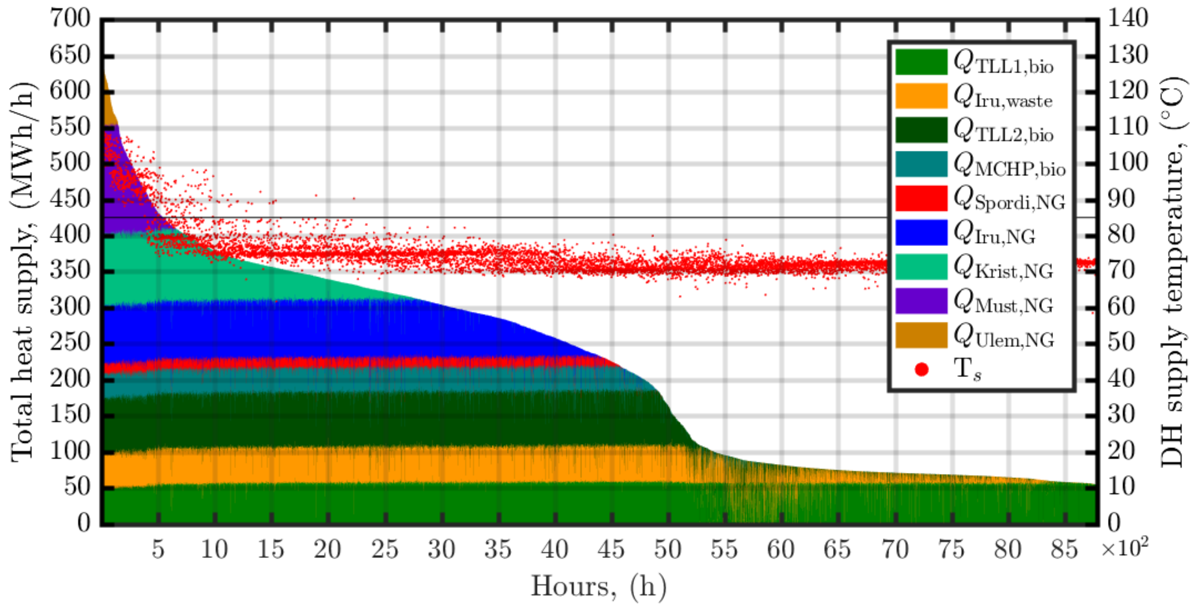

Figure 1: Load duration curve and DH supply temperature, based on data for 2016 [35,38]

It shows that the hourly peak demand was 660 MW, but this can be up to even 100 MW higher for very cold periods [38]. The baseload was covered by the biomass and waste CHP plants. The remaining heat, produced by the natural gas boilers, was used to determine the potential hourly heat supply by HPs. The potential heat supply of HPs was hence limited to 4500 hours per year for small HP capacities and would decrease for larger capacities, due to the characteristic shape of the load duration curve. Considering the temperature constraint for HPs, the maximum heat supply was reduced to 4000 h.

3.1. Potential locations and heat sources for HPs in Tallinn DH

Different locations, potential heat sources, capacity limitations and distances to the existing DH network were identified, which may be seen in Figure 2 and Table 2 and are explained in the following subsections. It was decided to focus on waste heat and natural heat sources. Industrial excess heat was not considered. Furthermore, it was decided to use available space and buildings at owned property from the utility company, which would reduce the investment costs. Further, a case #13 was added, which considered a groundwater HP located up to 100 m from the DH network anywhere in the city without potential savings on the property and building.

It was assumed that for HPs located near a CHP plant, the capacity fee for the transport of electricity could be avoided, because the CHP plant could supply the HP directly. Furthermore, it was assumed for case #12 that 2/3 of the 30 % heat loss in the 12 km transmission pipe could be avoided, if the HP was placed close to the area of Maardu.

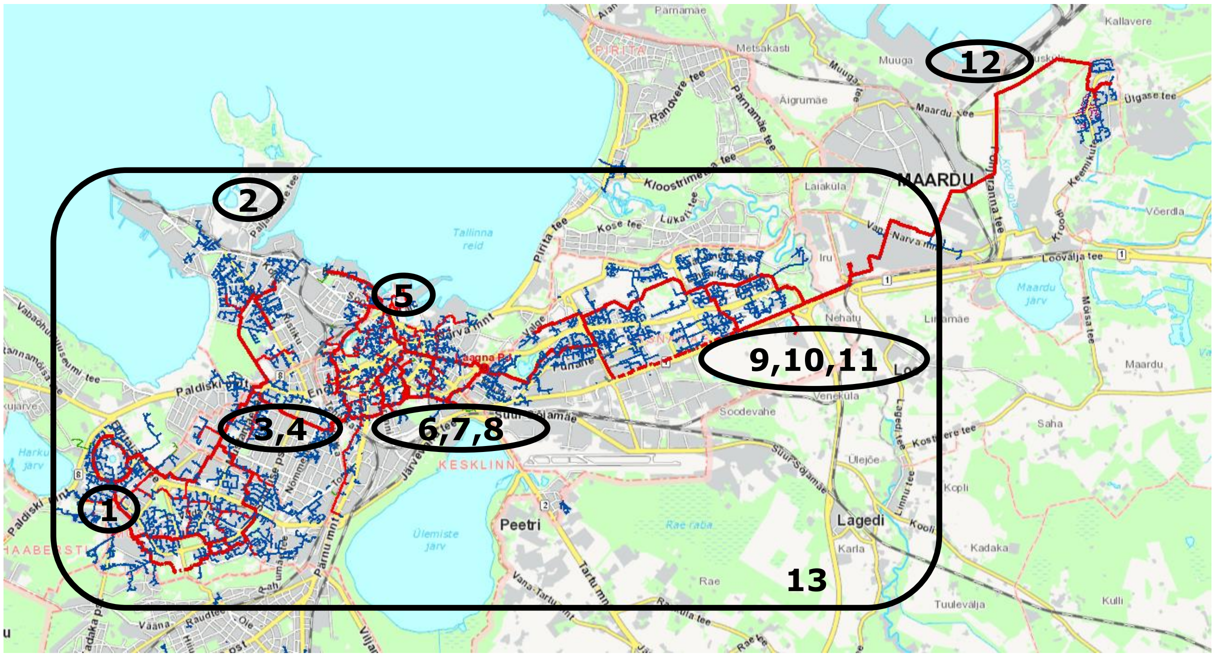

Figure 2: Possible locations for large-scale HPs and DH network of Tallinn

#	Location	Heat source	Limitation	Distance to DH (km)
1	Biomass CHP	Groundwater	2 MW	0
2	Sewage plant	Sewage water	4000 – 14000 m3/h	2.5
3	Boiler house	Ambient air	10 MW	0
4	Boiler house	Groundwater	1 MW	0
5	City centre	Seawater	No limit	0.2
6	Boiler house	Lake water	1200 m3/h	0.3
7	Boiler house	Groundwater	1 MW	0
8	Boiler house	Ambient air	10 MW	0
9	Biomass CHP	River water	6000-25000 m3/h	0.6
10	Biomass CHP	Ambient air	24 MW	0
11	Biomass CHP	Groundwater	6 MW	0
12	Maardu	Seawater	Local heat demand	2.0
13	100 m from DH	Groundwater	6 MW each	0.1

Table 2: Possible locations for large-scale HPs, limitations and distance to DH network

3.1.1. *Design conditions of heat sources and heat sink*

The design conditions for the different heat sources are shown in Table 3. A lower temperature difference was assumed for surface water to avoid freezing problems and to dimension the equipment accordingly.

Heat source parameter	Unit	Air	Groundwater	Seawater	River	Lake	Sewage
Inlet temperature	°C	-21	7	4	4	4	7
Temperature difference	K	6	6	3	3	3	6

Table 3: Heat source inlet temperatures for design conditions

The design temperatures of the HP heat sink were 85 °C for the supply temperature, and 40 °C for the return temperature. The forward temperature corresponds to the limit for the supply temperature given by pressure and temperature limits of state-of-the-art ammonia compressors for HPs [39].

3.1.2. *Ambient air*

Ambient air was analysed by use of hourly data for 2016 [38]. By considering known dimensions of existing evaporators installed at two large-scale air-source HPs in Denmark [40], the technology was limited by the available space at the boiler houses (#3 and #8). At the CHP plant (#10), dry coolers are already installed for cooling purposes in summer. It was assumed that they could be used in winter as heat source for the evaporators of the air-source HP, since they contain a glycol-water mixture. The installed capacity was converted for the heat source, using Eq. (9) and considered as the limitation. It was further taken into account that 25 % of that capacity is not used in winter, because of de-icing. Using the existing equipment would result in a cost reduction of the investment for case #10, as shown in Table 5.

3.1.3. *Groundwater*

In Estonia, the groundwater temperature at a depth of 25 m to 75 m is 6.5 °C to 7 °C [41–43]. A constant value of 7 °C was assumed. No additional details about the groundwater availability, usage and flows for the location of Tallinn were known. The largest HPs based on groundwater in Denmark have a capacity of 4 MW [31]. This requires a large amount of groundwater, which is difficult to extract and reinject without compromising the long term stability. This is why the practical limit for groundwater HPs may be at around 5 MW to 6 MW [44]. Furthermore, an analysis for an area in Copenhagen, Denmark, has shown that pumping 50 m^3/h of water influences the groundwater level temporary by 0.5 m at a distance to the pumping location of approximately 900 m [45]. Therefore, the limits of the groundwater HP capacities were estimated between 1 MW and 6 MW, according to the size of owned property for the cases #1, #4, #7, #11 and #13.

3.1.4. *Seawater*

Seawater temperatures are measured around the coastline of Estonia by the Estonian Weather Service [46]. Hourly data for 2016 were provided for the station in Pirita for a depth of 1.5 m. The seawater temperature may be more constant and a few degrees Celsius warmer during winter at lower depths. However, the seawater near Tallinn has a depth of approximately 10 m, which means that large temperature increases in winter were dismissed. The minimum measured seawater temperature was −0.1 °C, which is close to the freezing point of seawater in the Baltic Sea, which was estimated to be −0.5 °C, due to the salinity content of 15 ‰ to 25 ‰ as reported in [47]. This would limit the heat that can be extracted based on a design with a certain heat source temperature difference between inlet and outlet of the evaporator, as mentioned in Section 2.3.

3.1.5. *River water*

The Pirita River flows near the CHP plant east of Tallinn (#9). The monthly average water temperatures and minimum volume flow rates - obtained from [48] - were used as input and limitation on the heat source side. Also here, a minimum allowable water temperature out of the evaporator was assumed, which would limit the potential heat to extract during periods with very cold water temperatures.

3.1.6. *Lake water*

The biggest lake near Tallinn is Lake Ülemiste. One existing boiler house is located 300 m away from the lake. A HP could be placed here using surface water as heat source (#6). The lake is used as the largest water reservoir for Tallinn with 88 % of the water supply coming from that lake [49]. Therefore, changes in water temperature or contamination by any means are not desired. A potential use as heat source was still investigated by allowing a maximum cooling of the lake by 2 K as a result of the returning water of the HP. The water volume was calculated based on the surface area and average depth resulting in approximately 23.6 million m^3. Assuming a maximum operation of 2500 h and a cooling of the water by 3 K, the average volume flow rate for the heat source could be 1200 m^3/h. This limits the capacity of the source to 6 MW. Average monthly water temperatures were used based on a lake with similar characteristics found in [48] and a minimum water temperature was also considered.

3.1.7. *Sewage water*

A large sewage water treatment plant is located northwest from Tallinn, which could serve as potential heat source for HPs. Since the biological treatment process of the waste water is sensitive to changes in temperature, it was decided to investigate using the cleaned sewage water after the cleaning process as heat source, before it is sent to the sea [50]. The utility company AS Tallinna Vesi provided daily data of temperature and volume flow rate for one year [51]. The volume flow rate was only measured for the untreated water. However, the same volume flow rate was assumed for the cleaned water, even though this is not the case in reality. The treated water is pumped and stored in separate buffer tanks, afterwards pumped in another well and later self-flowing into the sea [50]. The daily volume flow rate was evenly distributed over the hours.

3.2. Economic input parameters

The O&M costs of HPs were based on [22,31]. The correlations for the investment costs for the HPs were based on Pieper et al. [23]. An overview of both is shown in Table 4. The investment costs are based on linear correlations with respect to the installed heating capacity. Equipment costs and building costs could be saved for the installation of HPs at some locations, which were deducted as shown in Table 5. Due to the location in the city centre, the construction costs for case #5, were set twice as high. No correlations were available for large-scale HPs using seawater, river water or lake water. The same costs as for a sewage water HP were assumed, since the installation may require similar type of equipment.

Parameter	Unit	Air	Groundwater	Seawater	River	Lake	Sewage
Investments, fixed	T€	188	505	484	484	484	484
Investments, variable	T€/MW	677	640	550	550	550	550
O&M, fixed	€/MW/yr	2000	2000	2000	2000	2000	2000
O&M, variable	€/MWh	1.0	2.0	2.0	1.3	1.3	1.3

Table 4: Investment and O&M costs based on [22,23,31]

Savings	T€	T€/MW	Applicable case #
Construction	21.77	84.31	1, 3, 4, 5, 6, 7, 8
Heat source	0.06	127.38	10

Table 5: Savings and additional costs for investment costs of HPs based on [23]

The investment costs of installing DH pipes was known for different nominal pipe sizes [35]. A linear correlation between costs and volume flow rate was developed based on a maximum velocity of 2 m/s, similar to Eq. (3). The required pipe diameter and the resulting heating capacity were then calculated. The cost correlations for installing DH pipes inside the city centre of Tallinn and further out are shown in Eq. (18) and Eq. (19), respectively. Eq. (19) was used for cases #9 and #12. For case #6, the heat source volume flow rate was used as dimensioning parameter for the costs, since the HP would be placed at the boiler house and not directly at the lake.

$$C_{\mathrm{DH,TLL}} = 478 + 0.79\,\dot{V}_{\mathrm{DH}} \qquad (€/\mathrm{m}) \tag{18}$$

$$C_{\mathrm{DH,rural}} = 283 + 0.56\,\dot{V}_{\mathrm{DH}} \qquad (€/\mathrm{m}) \tag{19}$$

The lifetime of the DH pipes and the HPs were assumed to be 30 years and 20 years, respectively [31,35]. A discount rate of 4 % was applied for calculating the annualized investment costs.

The optimization was performed from a private economic perspective, including energy taxes and transmission fees for electricity. Hourly day-ahead electricity prices for 2016 for Estonia from Nord Pool [52] as well as the current taxes and tariffs for Estonia/Tallinn from 2018 were applied, as shown in Table 6. An estimate for the transmission costs was made, since these costs depend on the kind of consumer, considered here to be the DH supply company located in Tallinn. It was assumed that the CHP plants could provide the electricity directly for the HPs located nearby. Therefore, the transmission fee was neglected for cases #1, #9, #10 and #11.

Parameter	Value	Unit
Nord Pool annual average price for 2016	33.06	€/MWh
Energy tax	4.47	€/MWh
Renewable energy tax	8.90	€/MWh
Transmission fee	18.63	€/MWh
Total	65.06	€/MWh

Table 6: Electricity taxes and tariffs for Tallinn

The price of DH is regulated in Estonia and is limited for each larger production unit, which will be evaluated regularly by the Estonian Competition Authority. The production price includes the LCOH, considering costs for fuel, electricity, depreciation, salaries and maintenance. The LCOH for one of the natural gas boilers used was 46.85 €/MWh for winter 2018 [53]. An overview of the input parameters for the natural gas boiler, which was replaced by HPs, may be found in Table 7. In addition, the emission factors are given for natural gas and the national electricity mix of Estonia [54,55]. The factor for electricity is rather large compared to values found in other countries, e.g. Denmark: 0.2 tonCO_2/MWh_{el}. The main reason for it is the use of power plants based on oil shale, which amount to a share of 80 % of the gross electricity generation in 2016 [56].

Parameter	Value	Unit
$LCOH_{NG}$	46.85	€/MWh
Boiler efficiency	0.93	-
CO_2 emissions of NG	0.198	tonCO_2/MWh
CO_2 emissions of electricity	0.95	tonCO_2/MWh_{el}

Table 7: Input parameters for natural gas boiler and emission factors based on [53–55]

3.3. *Sensitivity analysis*

A sensitivity analysis was performed by increasing the hourly electricity price from Nord Pool by 20 €/MWh in order to see how the economic parameters of the initial found optimum would change. This would show how sensitive the found solution is to changes in electricity price, which are difficult to predict over the lifetime of the plant.

In addition, the investment costs were increased by 30 % to see how the new found solution would differ from the initial found solution in terms of optimum HP capacities and heat sources as well as economic parameters.

4. *Results*

The results of the optimization are presented by showing the determined temperature profiles and Lorenz efficiencies for different heat sources as well as the optimum HP capacities, heat sources to be used and HP operations found by the model. These are shown together with technical, economic and environmental parameters.

4.1. *Heat source temperatures*

The temperatures of the different heat sources are shown in Figure 3. As it may be seen, the temperatures vary over the year for most heat sources, except for groundwater. In winter, sewage water and groundwater have the highest temperatures, followed by lake water, river water and seawater, and lastly at the lowest temperatures ambient air. The trend is different during summer. However, due to the existing baseload, little or none HP operation was expected during summer.

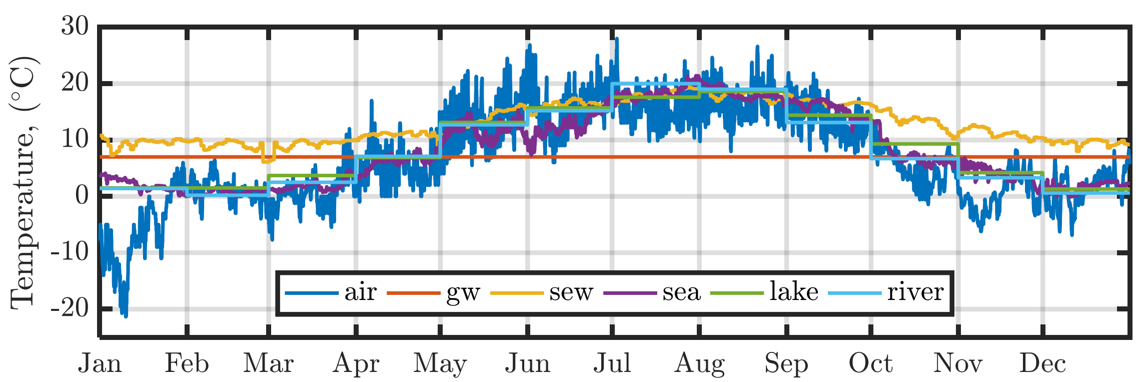

Figure 3: Heat source temperatures

4.2. *Lorenz efficiency of HPs over the year*

The Lorenz efficiency was calculated for each heat source and for every hour, taking changes in temperature of heat source and heat sink into account, as shown in Figure 4 and Table 8. It may be seen that the Lorenz efficiency η_L varies over the year and for the different heat sources. The values vary the most for ambient air, i.e. between 0.50 and 0.61. For surface water, the variation is between 0.53 and 0.60. For groundwater and sewage water the Lorenz efficiency varies less. This shows that the Lorenz efficiency may depend on the heat source temperature.

Furthermore, the Lorenz efficiency is higher during the winter months, with colder heat source temperatures and higher DH supply temperatures, than in summer, which may also be seen from Eq. (7). A higher Lorenz efficiency does not mean that the COP is also higher, since the Lorenz COP_L may be higher for the other heat sources. The results indicate further that assuming a constant Lorenz efficiency over the entire year and the same value for every heat source may impact the results of the optimization, if the heat source temperature varies considerably.

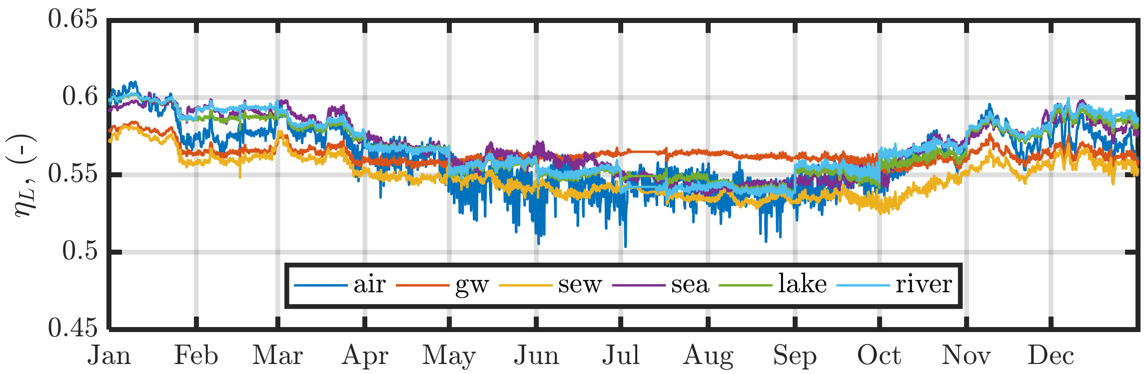

Figure 4: Lorenz efficiency over the year of HPs based on different heat sources

Parameter	Air	GW	Sew	Sea	Lake	River
Minimum hourly Lorenz efficiency	0.50	0.55	0.52	0.54	0.54	0.53
Maximum hourly Lorenz efficiency	0.61	0.58	0.58	0.60	0.60	0.60

Table 8: Minimum and maximum hourly Lorenz efficiencies

4.3. *Optimization results*

The results of the optimization are shown in Table 9. It was found to be optimal to install 122 MW HP capacity for the DH network in Tallinn consisting of sewage water (46 MW), river water (31 MW), ambient air (24 MW), seawater (13 MW) and groundwater (6 MW and 2 MW) HPs. The table also compares the results to typical values for Danish conditions.

The SCOP was 3.3, while it varied between 3.0 and 3.4 depending on the heat source. The number of FLH was 2670 h, which is lower than the maximum potential of 4000 h. The total annual production of heat by the HPs was 325 GWh. Compared to the total heat production of Tallinn by DH, the HPs would reach a share of 16 %, while the share of natural gas would be reduced from 50 % to 34 %.

However, this does not mean that the heat supply will become more sustainable, which is shown by the carbon-ratio. Since the carbon-ratio is above one, more CO_2 emissions would be emitted by the HPs compared to natural gas boilers. This happens, because the national electricity mix in Estonia has a quite high emission factor for CO_2. If the CO_2 emission factor of electricity would decrease from 0.95 $tonCO_2/MWh_{el}$ to 0.7 $tonCO_2/MWh_{el}$, the HPs would become more sustainable than using natural gas boilers. Using similar technology assumptions for a Danish case, the carbon-ratio was found to be 0.3.

Parameter	Unit	#1	#2	#9	#10	#11	#12	Total	Denmark [31]
Heat source		GW	Sew	River	Air	GW	Sea	Mix	
HP capacity	MW_{th}	2	46	31	24	6	13	122	
Heat production	GWh_h/yr	7	114	80	68	21	35	325	

Power consumption	GWh_{el}/yr	2	34	24	22	6	11	99	
Seasonal COP	-	3.3	3.4	3.3	3.0	3.3	3.3	3.3	3.5 to 4.0
Full load hours	h	3466	2495	2556	2814	3479	2810	2670	3000 to 6000
Levelized cost of heat	€/MWh_{th}	31.2	38.5	31.5	30.5	29.7	37.3	34.3	42 to 49
Cost of heat excluding investment	€/MWh_{th}	15.0	20.5	15.5	16.5	15.0	20.8	18.0	25 to 32
Cost of electricity	€/MWh_{el}	47.0	66.4	48.2	47.4	47.0	66.2	56.0	90 to 100
Net present value	M€	1.4	11.9	15.9	14.5	4.7	8.8	57.1	
Simple Payback Period	yrs	7.2	9.6	7.2	6.5	6.5	6.6	7.7	4 to 8
Carbon ratio	-	1.4	1.3	1.4	1.5	1.4	1.4	1.4	0.3

Table 9: Results of optimization

The costs of electricity differ for some of the heat sources, since the transmission fee could be avoided for the HPs located near the CHP plants (#1, #9, #10 and #11). This also has an impact on the costs of heat for the individual plants. Therefore, the HPs located near CHP plants were preferred for the operation due to the cost savings, even though they might not be the most efficient ones, see SCOP for sewage water HP. The HPs were operated according to their operational costs for every hour, after the investment was made. The levelized costs of heat were calculated as 34.3 €/MWh on average. This number corresponds to the minimum price the HP owner would need to pay off O&M and investment costs over the lifetime of the plants. This was 12.6 €/MWh cheaper than the levelized costs of using the natural gas boiler. The operational cost itself, considering O&M costs and electricity costs, was 18.0 €/MWh. The remaining part of 16.3 €/MWh accounted for the investment costs of the HPs and the DH piping.
If the HP owner would be able to receive the same amount for selling the heat as they would get by using the natural gas boiler, the simple PBT would be 7.7 years and the NPV with discounted cash flows would be 57.1 million € for the entire investment.
The costs of heat found through the optimization were lower than the ones currently found in Denmark [31], which was mainly because of the lower costs for electricity. The ambient conditions in Estonia are colder than the case for Danish conditions and the DH supply temperatures are higher, resulting in lower COP for the HPs than in Denmark. Considering the lower electricity costs, the PBT is similar as for Danish cases.

4.4. Load duration curve of HPs

The load duration curve of the HPs is shown in Figure 5. It may be seen that the two groundwater HPs were almost always operated at full load, due to their low operating costs. However, for the remaining 500 h another heat source was preferred, due to changes in COP of the heat sources. It is further shown that the river water and also the seawater HP have a significant drop in heat supply during 1500 h, shown here from 1000 h to 2500 h. This is due to the low water temperatures in February and December, as shown in Figure 3. The four HPs using groundwater (two locations), river water and ambient air were preferred due to their low operating costs, resulting from avoiding the capacity fee of electricity.

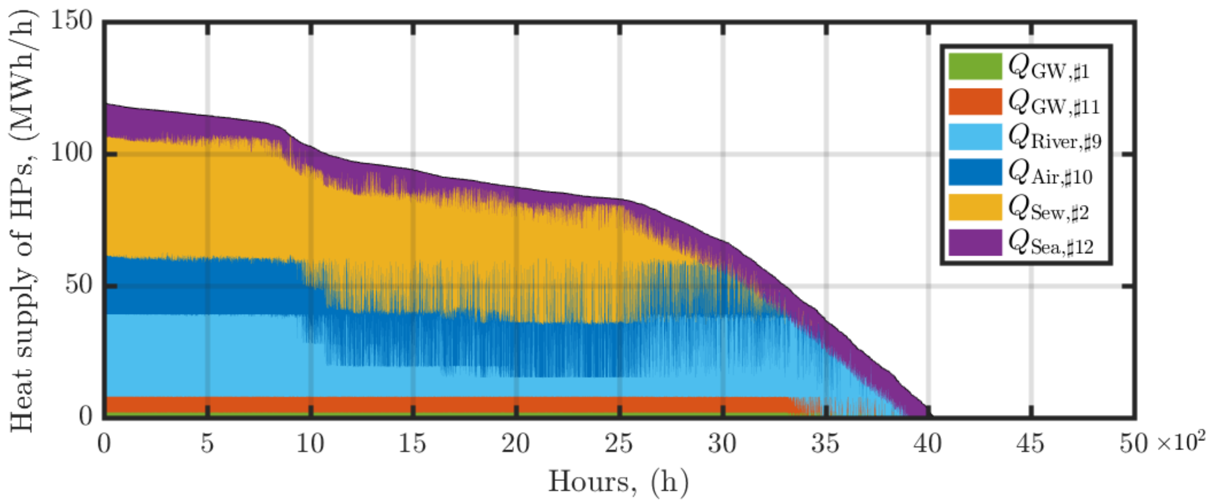

Figure 5: Load duration curve of HPs

4.5. Sensitivity analysis

The hourly electricity price was increased by 20 €/MWh in order to see how the economic parameters of the found optimum would change. The results shown in Figure 6 indicate increased costs of 14 % to 40 %, an increased PBT of 13 % and a reduced NPV of 44 %. Such an increase in electricity price is possible, since the average electricity price in 2018 increased to 46 €/MWh compared to 33 €/MWh in 2016 [52]. The found optimum would still result in a positive business case, which shows the robustness of the solution to changes on the electricity market.

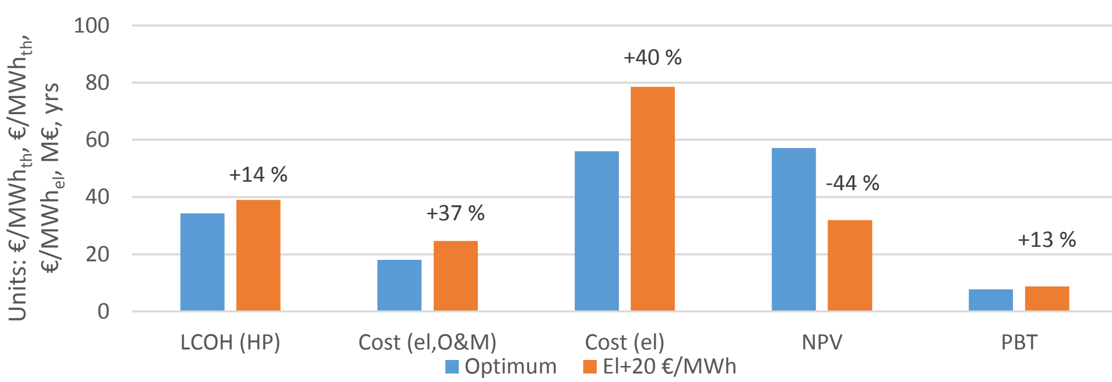

Figure 6: Change of economic parameters for increased electricity price

If the investment costs for the HP project as well as the DH piping costs were increased by 30 %, the same HP capacities would be proposed by the model apart from using river water as heat source. This heat source was completely removed from the new solution. The missing heat supply would be compensated by the remaining HPs, which resulted in higher utilization shown by an increase in FLH of 16 %, as shown in Figure 7.

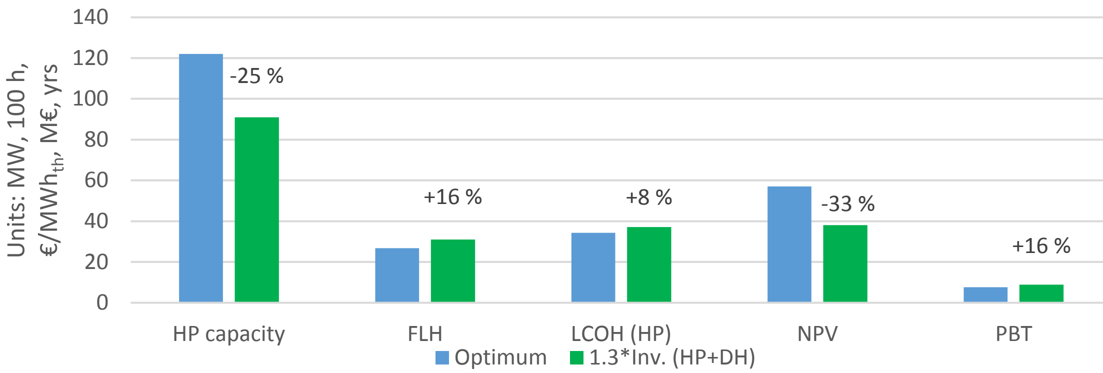

Figure 7: Change of parameters for increased investment costs

It may be further seen in Figure 7 that the LCOH increased by 8 % due to the increase in investment costs. The NPV decreases considerably by 30 %, while the PBT increases by 16 % to 8.9 years.

The change in operation of the HPs and how the HP based on river water was compensated is shown in Figure 8, which shows the load duration curves for the HPs based on the new solution with increased investment costs. Each of the HPs supplied heat nearly equivalent to full load operation for most of the time, except the seawater HP. This was due to the limitation in extractable heat from the heat source during periods with very cold seawater temperatures close to the freezing point. The heat supply decreased due to the lower temperature difference.

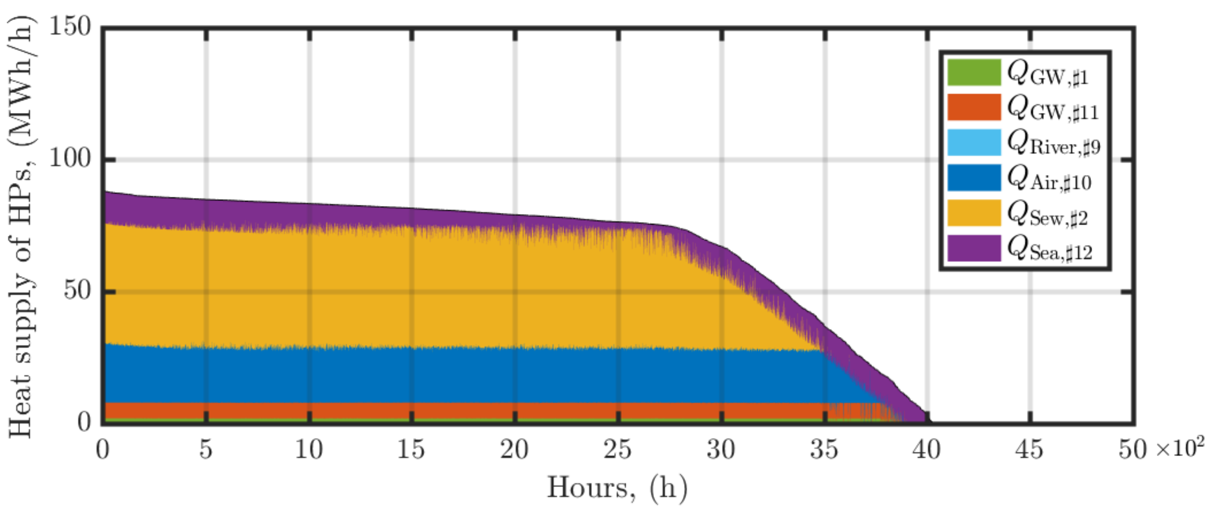

Figure 8: Load duration curve for increased investment costs of 30 %

Both the changes in electricity price and investment costs have shown that the found solution is robust to changes in costs. However, special care should be taken for the river water HP, since an increase in investment costs of 30 % would result in an unfeasible solution for this particular HP. It was found that the investment costs for the HPs and DH piping should not increase more than 15 % in order to be able to use river water as heat source. In this case, the HP capacity for using river water decreased to 21 MW and the NPV to 9.4 million €.

5. *Discussion*

The COP of the HP was calculated by an estimation method developed for design conditions of a single stage HP by Jensen et al. [15], which is an approximation. Even though the deviation in COP is expected to be small, it could have an impact on the results.

The Lorenz efficiencies calculated based on the COP estimation method varied depending on the variation in heat source temperature. Using this COP estimation method may therefore be a better approximation of COP than using a constant COP as in previous studies [3–8] or a cycle with constant Lorenz efficiency [10,11], if the heat source temperature varies considerably. This was also shown by Pieper et al. [57], who investigated the impact of using different COP estimation methods in energy planning.

Lorenz efficiencies between 50 % and 60 % were found, which is in the same range as found for existing large-scale HPs located in Denmark [58]. Improvements of the COP estimation method by expanding it for off-design conditions could result in smaller deviations compared to a thermodynamic model.

A constant isentropic compressor efficiency was assumed over the year. As it is shown in [32], the efficiency of compressors may vary for changes in pressure ratios because heat source and/or heat sink temperatures change. Such consideration could further improve the accuracy.

No sensitivity analysis of the input parameters found in Table 1 of the COP estimation method was performed.

In total, 122 MW of HP capacity was proposed for Tallinn, which has 400,000 inhabitants. The area of Greater Copenhagen has about 3 times more inhabitants and is suggested to benefit from a HP capacity in the magnitude of 300 MW by 2035 [59]. Bach et al. [60] investigated the possibilities of integrating large-scale HPs into DH for the area of Greater Copenhagen from a technical and private-economic perspective. The outcome of the study showed that 2500 to 3000 FLH of HP operation can be achieved for their investigated scenarios when connected to the transmission grid of the DH network. The presented study for Tallinn suggest 2670 FLH, which is thus in the same range.

A sensitivity study not included in this paper considering inclusion of a thermal storage in the system showed that thermal short-term storage would decrease the levelized costs of heat slightly. This happened, because the HPs operation could be controlled better to varying electricity prices by decoupling the heat demand and the heat supply by a few hours. This is in agreement with other studies about the inclusion of thermal storage, e.g. [61]. An impact on the optimal choice of heat source and HP capacity was not found. Therefore, thermal storage was not included in the analysis.

The proposed HP capacities and heat sources were based on the assumption that the capacity fee could be avoided for HPs built next to a CHP plant. If that is not applicable, the optimum solution may look differently.

A large share of the proposed HP capacities would be concentrated at two existing biomass CHP plants (61 MW, #9, #10, and #11). This could result in hydraulic issues or capacity limitations, when a large part of the production units is located at one location. On the other hand, the sewage water treatment plant is located at a very different location, and a 46 MW HP placed there could counterbalance such issues. Alternatively, further constraints could be applied to the optimisation problem in order to take capacity limitations into account.

The results showed that treated sewage water can serve well as a heat source for large-scale HPs, because the temperature of the water is higher than other available low-temperature heat sources. This was also stated by [19]. A size of 46 MW seems reasonable and possible, since a HP of 40 MW also using treated sewage water as heat source and ammonia as natural refrigerant was recently built in Sweden [62]. Another example of a large-scale ammonia HP exists in Drammen, Norway, which has a capacity of 13 MW, a COP of 3.05 and uses seawater as heat source [63].

The CO_2 emissions were calculated based on the national electricity mix, which would currently result in higher CO_2 emissions if HPs were installed compared to existing natural gas boilers. As calculated, the CO_2 emission factor of electricity would have to decrease from 0.95 $tonCO_2/MWh_{el}$ to 0.70 $tonCO_2/MWh_{el}$ for HPs to have lower greenhouse gas emissions than natural gas boilers. In 2014, 14 % of Estonia's electricity supply came from RES. In 2017, this value increased to 16.9 % [64]. In Estonia's National Development Plan of the Energy Sector until 2030[65], it is stated that the share of RES in the final electricity consumption shall be above 50 % in 2030. Therefore, it is expected that the CO_2 emission factor will further decrease, so that HPs will become more sustainable within the next decade.

The CO_2 emission calculations could look differently, if CO_2 emission factors were known on hourly basis or for Tallinn only. The electricity for the HPs connected to the biomass CHP would have zero CO_2 emissions. However, if this had been assumed, the national factors would have been affected negatively.

The investment and O&M costs of HPs were based on Danish conditions. Some of the costs might be different when applied to Estonia, such as construction costs. Other costs like the HP itself may be similar, however some offsets can apply to individual contributions. The sensitivity analysis has shown that the results to a large extent are robust to an increase in investment costs.

6. *Conclusion*

A novel modelling framework was presented, in which energy planning was combined with seasonal variations of heat source temperatures and HP COP as well as capacity limitations of the heat sources and technical constraints were taken into account. Using this modelling approach, the most suitable heat sources and optimal HP capacities were identified for the best economic solution of integrating large-scale HPs into the DH network of Tallinn.

Based on the results, it is recommended to install 122 MW HP capacity based on sewage water (46 MW), river water (31 MW), ambient air (24 MW), seawater (13 MW) and groundwater (6 MW and 2 MW). The HPs would be able to supply 16 % of the total heat demand of Tallinn. The HPs would currently result in more CO_2 emissions than providing the same heat with natural gas boilers, because the national electricity generation depends heavily on oil shale. However, considering the national development plans of Estonia's energy sector, the electricity supply in 2030 will be based on more than 50 % of RES. Consequently, HPs will then be a more sustainable solution than natural gas boilers. The LCOH based on HPs were 27 % cheaper than if the existing natural gas boilers were used. The sensitivity analysis showed that the solution is robust to changes in electricity prices and investment costs.

The results of applying the novel modelling approach showed that considerations of capacity limitations and seasonal variations of HP characteristics are important during the energy planning phase.

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Appendix 1 Comparison of COP calculations

Figure A. 1 shows hourly calculated COPs based on the COP estimation method [15] - considering the correction factor of 1.05 - and a thermodynamic model of a two-stage HP with open intercooler and ammonia as refrigerant. The thermodynamic HP model was developed in EES [66], with the same seawater temperatures as used for this study, constant DH supply and return temperatures of 60 °C and 35 °C, respectively and a polynomial to calculate the isentropic efficiency of a screw compressor [32]. Thereby, losses due to a mismatch between built-in pressure ratio and the actual pressure ratio were considered. The same input parameters were used for both models.

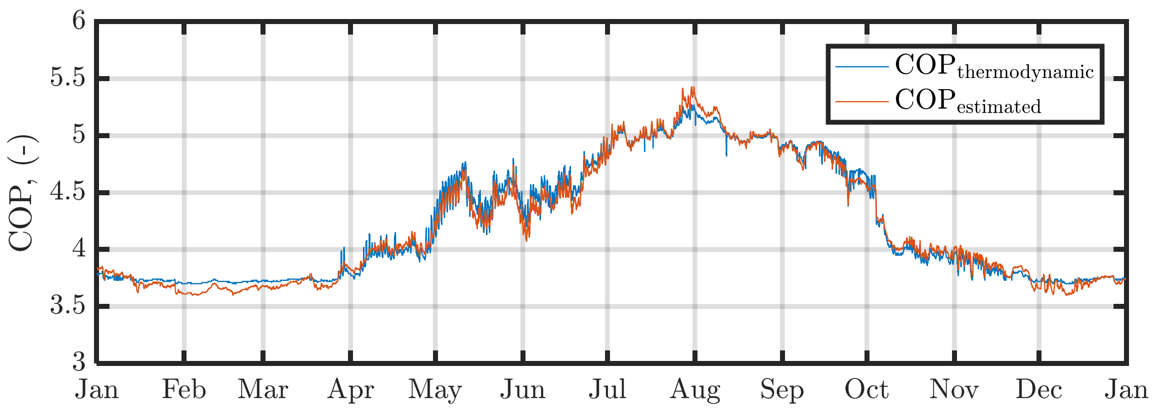

Figure A. 1: Hourly COP for thermodynamic model and COP estimation method